Renewable and Sustainable Energy Reviews 40 (2014) 11–27

Contents lists available at ScienceDirect

Renewable and Sustainable Energy Reviews

journal homepage: www.elsevier.com/locate/rser

On further enhancement of single-phase and flow boiling heat transfer in micro/minichannels

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ARTICLE INFO

Article history:
Received 25 September 2013
Received in revised form
24 June 2014
Accepted 19 July 2014

Keywords:
Microchannel
Heat transfer enhancement
Pressure drop
Bubble dynamics
Surface modification
Nanofluid
Nanoscale coating

ABSTRACT

With fast growing power consumption and device miniaturization, micro/minichannels are superior to macrochannels or conventional channels for high heat-flux dissipation due to their large surface area to volume ratios and high heat transfer coefficients. However, the associated large pressure drop penalty and flow boiling instability of micro/minichannels hinder their advancement in many practical applications. Therefore, enhancement techniques are required to stabilize the flow and further augment the heat transfer performance in micro/minichannels. This work first presents the classification of micro/minichannels for single-phase flow and flow boiling and gives a general statement of heat transfer enhancement. Then a state-of-the-art overview of the most recent enhancement techniques is specifically provided for further sing-phase flow and flow boiling enhancement in micro/minichannels. Two promising enhancement techniques, i.e., interrupted microfins and engineered fluids with additives are discussed for single-phase flow. For flow boiling, the focus is given on several selected enhancement approaches which can effectively mitigate flow boiling instability and another hot research topic, i.e., nanoscale surface modification. Besides, effects of wettability on bubble dynamics are presented, and a concept of flow-pattern based heat transfer enhancement is proposed. For both single-phase flow and flow boiling enhancement, a special emphasis is on those enhancement techniques with high thermal performance and relatively low pressure drop penalty.

Contents

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http://dx.doi.org/10.1016/j.rser.2014.07.171
1364-0321/

1. Introduction

Since the pioneering work of Tuckerman and Pease [1], which exhibited for the first time a high heat-flux dissipation of up to 790 W/cm^2 by using microchannels, a lot of effort has been devoted to investigate single-phase and two-phase fluid flow and heat transfer characteristics in microchannels and minichannels (micro/minichannels). Excellent reviews on this evolving topic have been reported in the literature, e.g., Thome [2], Celata et al. [3], Ghiaasiaan [4], Kandlikar [5] and Salman et al. [6].

Microchannels and minichannels (micro/minichannels) are preferred in diverse energy and process systems including compact heat exchangers, refrigeration and cryogenic systems, power electronics, automotive and aerospace industries, catalytic reactors, and fuel cells etc., mainly due to the following three reasons. First, in the scaling-down from macro to microscale, the volume decreases with the third power of the characteristic linear dimensions, while surface area only decreases with the second power. Therefore, micro/minichannels have relatively larger surface area to volume ratios which enable higher heat transfer rates than macrochannels or conventional channels. Second, fast fluid acceleration and close proximity of the bulk fluid to the wall surface in micro/minichannels give high heat transfer coefficient values. For single-phase laminar flow, a constant Nusselt number implies that the single-phase heat transfer coefficient is inversely proportional to the hydraulic diameter. In addition, compactness and high heat-flux dissipation are required as the scale of the devices becomes small while the power density becomes large.

1.1. Classification of micro/minichannels

A tricky aspect in micro/minichannels is how to identify the conventional-to-micro/miniscale threshold. That means, how small a channel can be called a micro/minichannel and accordingly its behavior starts to deviate from the predictions of conventional channels. Several criteria have been given in the literature [7–9]. Among them, the conventional-to-micro/minichannel threshold developed by Kandlikar and Grande [7] by arbitrarily adopting an absolute hydraulic diameter of 3 mm was commonly used in many previous studies for simplicity. However, this criterion does not consider the effect of the channel size on physical mechanisms and the differences between conventional channels and micro/minichannels, especially for flow boiling. Based on this, Li and Wu [10] generalized a conventional-to-micro/minichannel threshold for flow boiling as follows for both normal gravity and microgravity conditions:

$$Bo \times Re_l^{0.5} = \frac{g(\rho_l - \rho_v)D_{th}^2}{\sigma}\left(\frac{G(1-x)D_{th}}{\mu_l}\right)^{0.5} = 200 \tag{1}$$

where Bo and Re_l are the Bond number and the liquid Reynolds number, respectively. The Bond number is a measure of the importance of body forces (almost always gravitational) compared to surface tension. The liquid Reynolds number gives a measure of the ratio of liquid inertia forces to liquid viscous forces. When $BoRe_l^{0.5} \le 200$, micro/minichannel phenomenon dominates and the rapidly produced bubbles are confined and elongated in the channel; when $BoRe_l^{0.5} > 200$, conventional channel theory can explain the experimental data. This criterion, i.e., Eq. (1), has been verified further in [11–13].

In the present study, we define the ranges for micro/minichannels for single-phase flow and flow boiling separately. For single-phase flow, we arbitrarily adopt the hydraulic diameters of micro/minichannels within the range of 100 μm $\le d_h \le$ 3 mm. The upper limit of 3 mm was adopted for single-phase flow based on the commonly used threshold by Kandlikar and Grande [7]. In addition, the hydraulic diameters in highly compact plate heat exchangers, plate-fin heat exchangers, and printed circuit heat exchangers are within the range of several hundred micrometers to several millimeters. For flow boiling, we set the hydraulic diameter range for micro/minichannels as 100 μm $\le d_h \le D_{th}$. The upper limit D_{th} for flow boiling can be calculated from Eq. (1).

A lower limit of a magnitude of 100 μm was chosen to ensure that no-slip boundary condition exists on channel boundaries and thus the continuum assumption is still valid, even for single-phase vapor or gas flow. Due to the small size of these channels, the length scale is comparable to the molecular mean free path (λ). Deviation from the continuum theory will happen. The non-dimensional Knudsen number ($Kn = \lambda/L$) is defined to analyze this deviation. Parameter L denotes an appropriate length scale of the channel. Hydraulic diameter can be used as L for Kn estimation. When the Knudsen number is in the range of $0.001 < Kn < 0.1$, a nonequilibrium state occurs very close to the wall, which is initiated by domination of molecular collisions with the walls over intermolecular collisions [14]. For air at ambient pressure and at room temperature (25 °C), λ is about 68 nm [15]. The Knudsen number for air is about 6.8×10^{-4} for a hydraulic diameter of 100 μm, which is less than 0.001. In general, the continuum assumption is still valid for single-phase vapor or gas flow when using 100 μm hydraulic diameter as the lower limit for micro/minichannels.

1.2. Heat transfer enhancement in general

Heat transfer enhancement is very important for various heat transfer systems as well as for energy conservation and environment protection. Enhancement is normally concerned with increasing heat transfer coefficient. The goal of enhancement techniques may be to reduce the size of the heat exchange equipment for a given duty, to increase the capacity of an existing heat exchange equipment, or to reduce the approach temperature difference [16]. Heat transfer enhancement techniques can be classified either as passive (no external power needed) or as active (external powered required). Bergles [17] defined the four generations of heat transfer technology using passive techniques. Most of the heat transfer enhancement techniques were covered in the classic book by Webb and Kim [18].

The mechanism of heat transfer enhancement is a strong function of the nature of the fluid stream (gas, liquid, liquid–gas or liquid–vapor mixture, in some cases solid, or a mixture of liquid, gas (vapor) and solid) and the mode of heat transfer (e.g., natural convection, forced convection, boiling, condensation, radiation, etc.).

The progress in micro- and nanoengineering opens new room for micro- and nanoscale enhancement techniques. Fig. 1 presents an illustration of surface passive heat transfer enhancement techniques at macro-, mini/micro- and nanoscales. For single-phase flow, macroscale and mini/microscale techniques can be used to enhance heat transfer by increasing the heat transfer area and/or modifying the velocity profile (to produce turbulence). Generally, nanoscale technique does not apply for single-phase

Nomenclature

Bo	Bond number, $g(\rho_l-\rho_g)d_h^2/\sigma$
D_{th}	flow boiling threshold diameter (m)
c_d	downstream radius of curvature (m)
c_u	upstream radius of curvature (m)
d_h	hydraulic diameter (m)
G	mass flux (kg/m^2 s)
g	gravitational acceleration (m/s^2)
h	heat transfer coefficient (W/m^2 K)
h_{lv}	latent heat of vaporization (J/kg)
k	thermal conductivity (W/m K)
N_a	nucleation site density (sites/cm^2)
Nu	Nusselt number (hd_h/k)
q	heat flux (W/m^2)
R	flow instability parameter
r	effective nucleation cavity size (m)
Re_l	liquid Reynolds number, $G(1-x)d_h/\mu_l$
T_w	wall temperature (K)
T_{sat}	saturation temperature (K)
x	vapor quality

Greek symbols

ΔT_w	wall superheat (K)
δ_t	thermal boundary layer thickness (m)
θ	contact angle (deg)
μ	dynamic viscosity (Pa s)
ρ	density (kg/m^3)
σ	surface tension (N/m)
τ	the net surface tension per unit area (N/m^2)
φ	volume concentration

Subscripts

ave	average
l	liquid
ONB	onset of nucleate boiling
sp	single-phase
v	vapor

flow due to the limited turbulence and available heat transfer area increase. However, nanoscale techniques such as nanocoating can modify the surface condition (hydrophilic vs. hydrophobic) which may affect the heat transfer behavior [19]. For nanoscale porous layers, the surface heat transfer area may increase by many times; however, the liquid or gas entrapped in the nanopores cannot move freely to benefit heat transfer.

All the three kinds of surface passive heat transfer enhancement techniques at macro-, mini/micro- and nanoscales, can be adopted for two-phase flow. Proper selection of macroscale fins

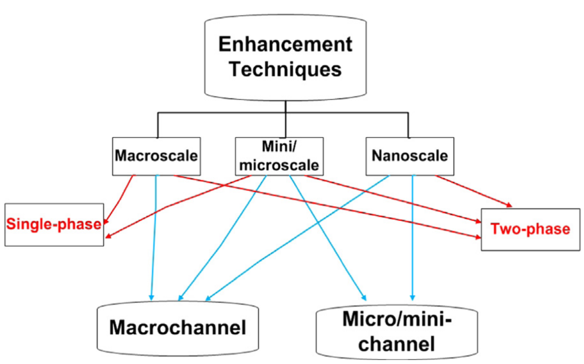

Fig. 1. Surface passive enhancement techniques for single-phase and two-phase flow.

can achieve filmwise condensation. The flow boiling or condensation enhancement by microfins (one kind of mini/microscale enhancement technique) is partly due to the mere increase in the effective surface area, and additionally to the turbulence induced in the liquid film by the presence of microfins and to the liquid drainage effect by surface tension [20,21]. Nanoscale modifications can change the wettability of the original heat transfer surface greatly. Wettability affects bubble dynamics, and thus heat transfer performance. Mini/microscale and nanoscale porous layers will add available nucleation sites and stabilize the nucleation process due to the interconnection of nucleation sites, improving flow boiling heat transfer. More specifically, Table 1 compares the typical advantages and disadvantages of the three kinds of surface passive techniques.

As shown in Fig. 1, the three kinds of surface passive techniques at macroscale, mini/microscale and nanoscale, e.g., fin, microfin and nanopillars can be adopted in macrochannels for possible heat transfer enhancement. Micro- and nano-features may give superior thermal performance when fabricated or deposited on the surfaces of micro/minichannels appropriately. Other conventional passive enhancement techniques such as fluid additives (e.g., nanoparticles, phase change material particles) [22,23] and active enhancement techniques like vibration and electrostatic fields [24] may also be beneficial for heat transfer enhancement in micro/minichannels.

Table 1
Typical surface passive heat transfer enhancement techniques.

Surface type for heat transfer enhancement	Main characteristics	
	Advantage	Disadvantage
Macroscale	Decent increase in surface heat transfer area; strong fluid mixing and secondary flow; very efficient for single-phase laminar flow; high heat transfer enhancement	Relatively high pressure drop penalty; decrease in flow area; not applicable for micro/minichannels
Mini/microscale	Restructuring the surface disturbing the boundary layers of single-phase flow or forming artificial nucleation sites for two-phase flow; very efficient for boiling and condensation with large increase in heat transfer coefficient; moderate pressure drop penalty	Moderate increase in surface heat transfer area; inefficient for single-phase gas flow
Nanoscale	Numerous nucleation sites; flow instability mitigation; suitable for spatially non-uniform power distribution; low pressure drop penalty	Very little increase in surface heat transfer area; inefficient for single-phase flow; fouling; relatively new and needs durability tests

Smooth micro/minichannels may not be sufficient for high heat-flux dissipation applications such as high power electronics. Thus conventional or advanced enhancement techniques which are suitable for micro/minichannels are required to be implemented in micro/minichannels for further heat transfer enhancement to meet the increasing cooling demand and the further device miniaturization.

1.3. Objectives of the present work

The present study mainly focuses on giving a state-of-the-art overview and suggests relevant future perspectives on conventional and advanced heat transfer enhancement techniques, especially the most recent enhancement methods, for further single-phase and flow boiling heat transfer enhancement in micro/minichannels. Flow boiling enhancement mechanism is also discussed briefly. A special emphasis is on those enhancement techniques with high heat transfer enhancement and relatively low pressure drop penalty.

2. Single-phase enhancement techniques for micro/minichannels

For single-phase flow in micro/minichannels of hydraulic diameters within the range of $100\ \mu m \le d_h \le 3$ mm, conventional fluid flow and heat transfer theories can be used to predict the hydraulic and thermal characteristics in micro/minichannels only when the following effects are appropriately accounted for: microchannel geometry, experimental uncertainties, entry and exit losses, surface roughness, entrance effects, conjugate heat transfer, viscous heating and temperature dependent properties. For hydraulic diameters less than about 100 μm, more scaling effects, such as electric double layer (EDL) effects, rarefaction and compressibility effects should be considered [25,26]. The above possible scaling effects, often negligible in conventional macrochannels, may now have a significant influence for microchannels [27].

Table 2 lists recent heat transfer enhancement techniques for micro/minichannels, which include passive enhancement techniques, e.g., surface roughness, reentrant cavities, dimples and protrusions, grooves, wavy/curved microchannels, surface nanostructures, vortex generator, supercavitating flow, interrupted microfins and additives, and active enhancement methods like synthetic jets. Besides, tree-shaped or fractal-shaped microchannel network based on the constructal theory may enhance heat transfer efficiently [64,65]. Most numerical and experimental studies given in Table 2 are focused on laminar-flow heat transfer enhancement. Turbulent flow generally entails higher pressure drop than laminar flow. High pressure drops not only require high pumping power but also pose leaking problems in micro heat sinks or microchannel assemblies. Therefore, heat transfer enhancement techniques need to be optimized for different working conditions to increase the heat transfer with low accompanied pressure drop penalty. Among various techniques to improve single-phase thermal performance in micro/minichannels, two approaches, interrupted microfins and additives are detailed. Micro/minichannels with interrupted microfins are very promising for high heat-flux dissipation. Engineered fluids with additives, very popular in research during the last ten years, may be an efficient way to significantly improve the thermal performance without large pressure drop penalty.

Table 2
Enhancement techniques for single-phase flow in micro/minichannels.

Enhancement techniques	Fluid/flow regime (*Re*)/possible enhancement mechanism	Remarks
Surface microstructures		
Surface roughness [28,29]	Water/laminar/disturbing boundary layer, possible eddy generation	Numerical simulation; the Poiseuille number for laminar flow strongly influenced by *Re*, relative roughness, and fractal dimension of the surface
Reentrant cavities (fan-shaped/triangular) [30]	Water/laminar (100–1100)/Interrupted boundary layers and repeated developing flow	Numerical simulation considering conjugate heat transfer; geometry parameters optimized partly with the criterion $Nu/Nu_o/(f/f_o)^{1/3} > 1.4$
Dimples/protrusions/oval dimples [31,32]	Water, air/laminar, turbulent/vortices shedding, flow reattachment, impingement	Numerical simulation; heat transfer enhancement with low pressure drop penalty; combined dimple+protusion better than dimple or protusion
Grooved surfaces (Rectangular, arc shapes) [33]	Water/laminar (700–1500)/repeated developing flow	Numerical simulation considering conjugate heat transfer; arc shapes perform better than rectangular shape; groove size and spacing optimized
Wavy/curved microchannels [34–36]	Water/laminar/secondary flow, enhanced fluid mixing, boundary layer thinning	Numerical simulation; heat transfer enhancement with low pressure drop penalty by using optimized parameters
Surface nanostructures nanowires [37]	Water/laminar (106–636)/enhanced wettability and slight surface area increase	Experiments; no obvious effect of fluid flow on the surface morphology of the nanostructures; heat transfer and pressure drop both increased about 25%
Vortex generator [38]	Water/laminar, turbulent (170–1200)/secondary flow, strong fluid mixing	Experiments; critical *Re*: 600–730; moderate heat transfer enhancement with large pressure drop for both laminar flow and turbulent flow
Supercavitating flow [39]	Water/laminar/transition flow patterns, rapid flow agitation and mixing, liquid jet	Experiments; significant heat transfer enhancement with minor pressure drop penalty; required large pressure drop to initiate cavitation
Interrupted fins: Micro pin-fins, cross-cut channels, offset strip fins, oblique fins, etc. [40–46]	Water, air etc./laminar, turbulent/repeated developing boundary layer, fluid mixing, secondary flow	Experimental and numerical studies; generally large heat transfer enhancement and high pressure drop penalty; streamlined fin shapes, e.g., S-shaped fins, and diamond-shaped fins, oblique fins achieved relatively high heat transfer enhancement with low pressure loss; promising for high heat-flux dissipation
Additives		
Slug-train coflow immiscible droplets [47,48]	Water, silicone oil, PAO/laminar/blockage effect, disturbing thermal layer	Numerical simulation; using a front tracking method to track the interface between droplets; heat transfer enhancement with moderate friction losses
Nanoparticle (metal, oxide, carbon nanotube, etc) [49–56]	Liquid (water, EG, PAO, etc) and nanoparticle mixtures/laminar, turbulent/thermal conductivity increase, etc	Experimental and numerical studies; controversy on anomalous heat transfer enhancement; mostly, heat transfer enhancement coupled with pressure losses; stability problems exist even for nanofluids
Micro/nano- encapsulated phase change materials (PCM) [57–61]	PCM fluids/laminar/enhanced heat capacity due to the PCM's latent heat of fusion upon melting	Experimental and numerical studies; PCM fluids perform better under constant heat flux condition; PCM increases the *Nu* and decreases the bulk mean temperature; viscosity of PCM fluid also increases
Synthetic jets [62,63]	HFE7300, R134a/laminar, turbulent/disrupting boundary layers and jet impingement	Experiments; adding net momentum flux into a stream without adding mass flux by synthetic jets; optimal operating range: transitional to weakly turbulent flow

2.1. Interrupted microfins

Single-phase flow in micro/minichannels can be enhanced effectively by replacing the traditional continuous channel walls with short fins. Short fin lengths are preferred to disrupt the thermal boundary layer and obtain repeated developing flow. The fin height is also controlled to maintain high fin efficiency. Staggered fin arrangement is adopted usually, so that the thermal boundary layers upstream and downstream of the fins do not affect each other. Flow mixing and flow uniformity are greatly improved by using interrupted microfins than straight plain microchannel arrays. In addition, deterioration in heat transfer due to axial conduction along the walls can be mitigated by interrupted microfins.

Interrupted fins can augment heat transfer up to several times and reduce the maximum wall temperature rise. Fin shapes, fin dimensions, and fin array parameters such as fin pitch and fin attack angle can largely affect the hydraulic and thermal characteristics of liquid or gas flow in micro/minichannels. Therefore, optimization of fin-related parameters is critical for application of interrupted fins in micro/minichannels. Different fin shapes (e.g., diamond-shaped interrupted microfin, offset strip fin, staggered micro pin fin, oblique microfin) have been investigated both experimentally and numerically in the literature, see Refs. [40–46] for details. Fig. 2a illustrates the plan view of an enhanced microchannel array formed by oblique microfins, along with the flow paths for the main flows and secondary flows. Fig. 2b plots the average Nusselt number versus Reynolds number for a conventional straight microchannel array and the enhanced microchannel array by oblique microfins. A 103% heat transfer enhancement is achieved at the Reynolds number of 780, when the average Nusselt number increases from 11.3 to 22.9 [43].

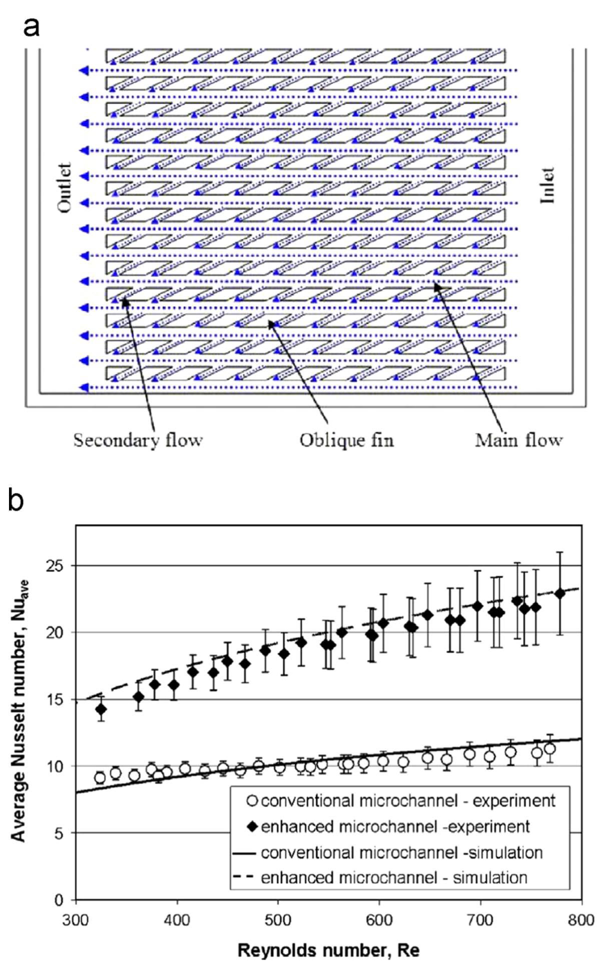

Fig. 2. (a) Illustration of an enhanced microchannel array by oblique microfins [43], and (b) average Nusselt number vs. Reynolds number for the enhanced microchannel array and a conventional straight microchannel array with a nominal channel width of 0.5 mm [43].

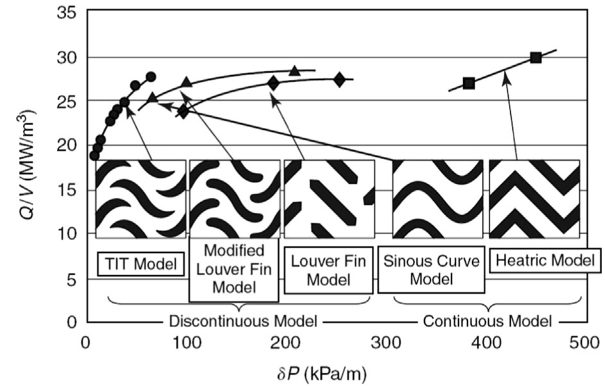

Fig. 3. Comparison of five flow passage patterns, Q/V is the heat transfer rate per unit volume [45].

To further reduce pressure losses, Tsuzuki et al. [45] developed discontinuous S-shaped fins and achieved a pressure drop only 20% of the conventional zigzag passage, as shown in Fig. 3. The S-shaped fins also performed better than wavy/curved microchannels and louvered fins. The S-shaped fin has an almost straight part in the middle and arc shaped parts in the head and tail. A continued investigation for S-shaped fins suggested that fin roundness at the head and tail edges of the fins minimally affect the heat transfer performance but greatly influence the pressure losses. Less fin roundness is required to reduce the pressure drop. Rounded fins with 0.1 mm radius increased the pressure drop by about 30% compared to the fin designed with no roundness [45].

A streamlined fin shape airfoil was adopted in a printed circuit heat exchanger model in the 3-D numerical analysis by Kim et al. [46], as shown in Fig. 4. Simulation results showed that the total heat transfer rate per unit volume of the airfoil-shaped fin arrays was almost the same as zigzag flow passages while the pressure drop was reduced to one-twentieth of that in zigzag flow passages. In the airfoil-shaped fin model, the enhancement of heat transfer area and the uniform flow configuration contributed to obtain the same heat transfer performance with zigzag channels. The pressure drop reduction in airfoil-shaped fins is caused by suppressing the generation of separated flow owing to the streamlined shape of airfoil fins.

2.2. Engineered fluids with additives

As given in Table 2, additives for single-phase flow in micro/minichannels include immiscible liquid droplets and solid particles. Nanoparticles and micro/nano-encapsulated phase change materials (PCM) are two common kinds of particle additives. Recently Fischer et al. [49] introduced slugs of immiscible liquid droplets into the base fluid to augment heat transfer. The Nusselt number can be increased by establishing an elongated droplet flow in the microchannel, as shown in Fig. 5. The suspended fluid in the elongated droplets is a nanofluid consisting of alumina nanoparticles and a solvent, polyalphaolefine (PAO), while the base fluid is water. The average Nusselt number is tripled compared to liquid flow, due to the pronounced recirculation inside and between the elongated droplets by blockage effects.

2.2.1. Nanofluids

Nanofluids are engineered colloidal suspensions of nanoparticles of a base fluid, which are more stable than microparticle

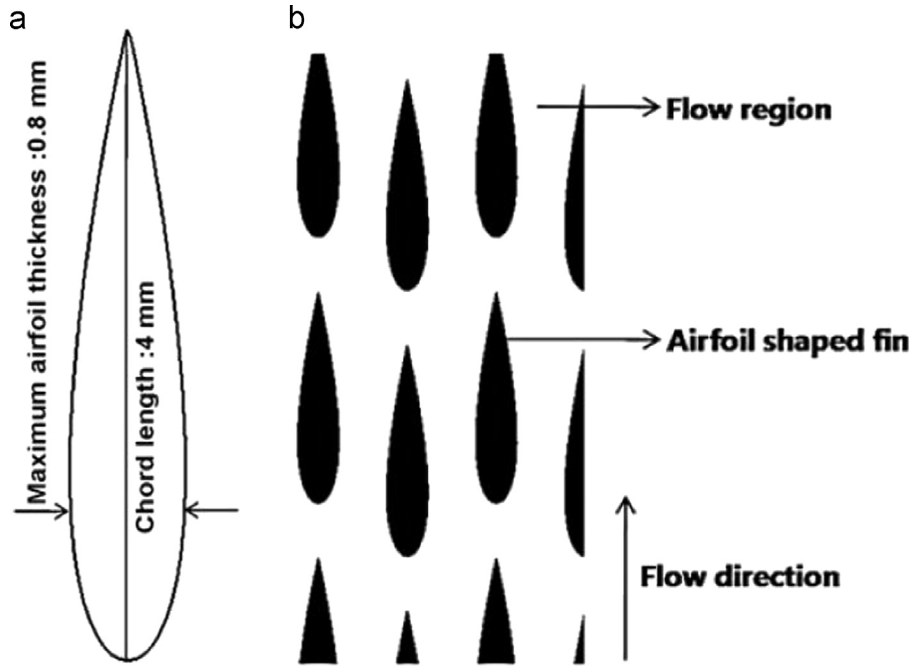

Fig. 4. (a) NACA 0020 airfoil shape and (b) channel configuration of airfoil fins [46].

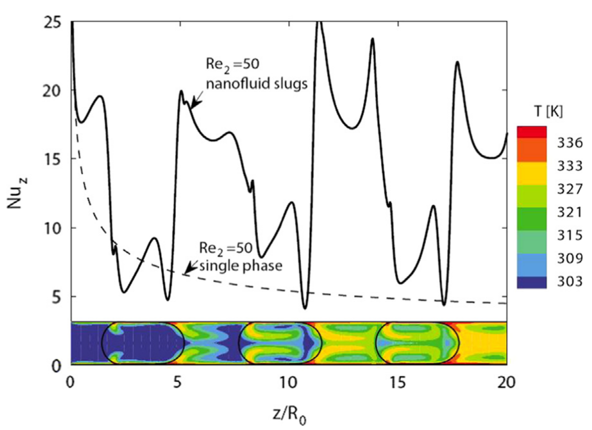

Fig. 5. Local Nusselt number and the temperature field in the axial direction for liquid–liquid flow with elongated droplets [47].

colloids, with less particle setting, channel erosion and clogging. Nanofluids generally provide higher thermal conductivity compared to their base fluids due to the relatively large thermal conductivity of the added nanoparticles. No agreement on anomalous heat transfer enhancement has been achieved up to now. Buongiorno and coworkers [66,67] concluded that there seems to be anomalous heat transfer enhancement in the entrance region in nanofluid laminar flow, while existing correlations can reproduce the turbulent convective heat transfer behavior of nanofluids accurately by adopting the measured temperature- and loading-dependent thermal conductivities and viscosities of the nanofluids in the analysis. Li et al. [68] obtained exact solutions for fully developed laminar flow in channels and tubes and concluded that: (a) the anomalous heat transfer enhancement was captured, especially for the case of titania-water nanofluids in a tube when the nanoparticle volume concentrations are larger than 2% and (b) the maximum Nusselt number based on the bulk mean nanofluid thermal conductivity is achieved at $N_{BT} \approx 0.5$, while it becomes lower than that of the pure fluid at $N_{BT} < 0.3$. The parameter N_{BT} indicates the ratio of Brownian and thermophoretic diffusivities.

For single-phase nanofluid flow in circular tubes, Lee and Mudawar [49] suggested that higher heat transfer coefficients can be obtained for laminar flow, while the enhancement was far weaker for turbulent flow, as shown in Fig. 6. The weaker enhancement for turbulent flow, as indicated by the Dittus–Boelter equation, was attributed to the weaker dependence of the heat transfer coefficient on thermal conductivity as well as decreased specific heat and increased viscosity with increased nanoparticle concentration. Besides, it is shown in Fig. 6 that the heat transfer enhancement of nanofluids also depends on the physical properties of the base fluid. Copper oxide-water nanofluids of particle diameters of 28.6 nm of different volume concentrations were shown to enhance heat transfer in a trapezoidal microchannel [53], as illustrated in Fig. 7. Volume concentrations of 1% and 4% Cuo–water nanofluids show about 15% and 20% average Nusselt number increases compared to pure water based on a constant Reynolds number comparison, respectively. However, as stated in Yu et al. [69] and Wu et al. [70], the constant Reynolds number basis can be misleading because the net result for the constant Reynolds number comparison is a combination of the nanofluid property effect and the flow velocity effect. Due to the higher viscosity of the nanofluid, the flow velocity in the nanofluid is generally higher than that of the base fluid at the same Reynolds number, which provides an advantage for the nanofluid compared to the base fluid. If the base fluid is to be pumped at the same flow velocity as the nanofluid, it may approach or exceed the thermal performance of the nanofluid. The result based on constant Reynolds number will be more misleading at higher estimated or measured relative viscosity values. As the viscosity increase is larger than the thermal conductivity increase for the CuO–water nanofluid, lower enhancement values will be obtained at a constant flow velocity basis than that shown in Fig. 7.

The associated viscosity increase by the nanoparticle additives will increase the pumping power, which will decrease the benefit of heat transfer enhancement. In addition, stability may still be a problem for nanofluids. Normally, surfactants are added for long-term use. Therefore, the effects of surfactants on the fluid flow and heat transfer also need to be considered. Besides, the high price of

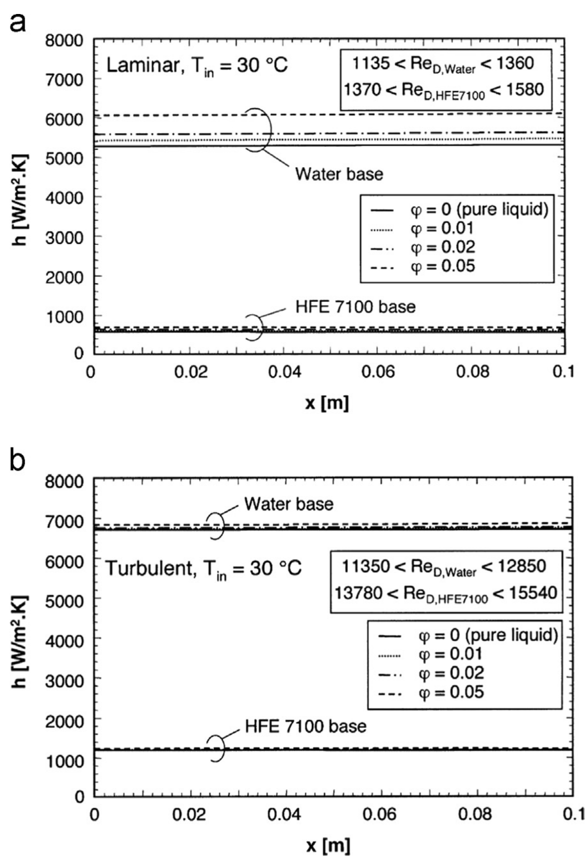

Fig. 6. Axial variations of heat transfer coefficient for (a) laminar flow and (b) turbulent flow at different alumina nanoparticle concentrations in water and HFE7100 base fluids [49].

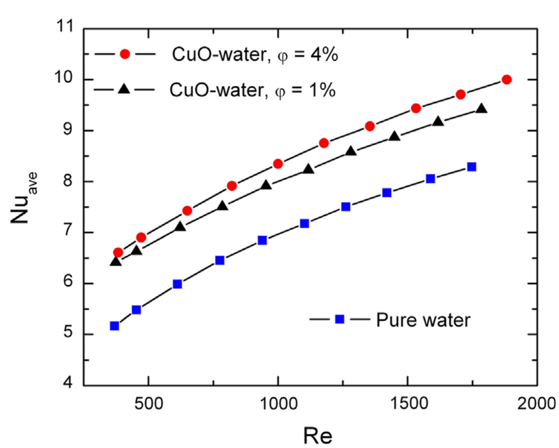

Fig. 7. Average Nusselt number vs. Reynolds number [53].

the nanofluid is one of the major barriers to apply nanofluids in a wide range of applications. Nanofluid has a relatively high cost due to the ultra-fine techniques and high accurate equipment used to synthesize and produce nanometer-sized particles. For microchannels already equipped with other enhancement techniques, effects of nanoparticles are still unclear. Probably, nanoparticles will present negligible enhancement when a large heat transfer augmentation has already been achieved by another enhancement technique.

2.2.2. *Phase change materials (PCM)*

Major factors that make phase change materials very attractive for thermal energy storage and thermal control are high energy storage density and small temperature variation. The phase change material is micro/nano-encapsulated and suspended in a base heat-transfer fluid to form phase change slurries. Encapsulating the phase change material in very small capsules is expected to eliminate any segregation during the phase change [71]. The capsule shell of the PCM is very thin. Paraffin wax, salt hydrates and Eutectics etc, can be used as phase change materials. Ideal PCMs for heat transfer enhancement should have high heat of fusion, high thermal conductivity, little volume change during phase change, narrow melting temperature range, chemical stability and good compatibility.

Micro/nano-PCM particles can be adopted as additives for liquid flow in micro/minichannels. Results from the previous numerical simulations [57–60] on PCM fluids have been promising with the main advantage of achieving lower wall temperature than conventional single-phase fluids under the same heat flux conditions. In these studies, the particle distribution was assumed homogeneous. Homogeneous distribution was verified by Kuravi et al. [72] for nano-encapsulated PCM slurries. Stratification and sedimentation can be minimized by using capsules of very small diameters and adopting PCM particles with density approximately equal to that of the suspending fluid [73].

Recently, Lenert et al. [57] simulated local heat transfer enhancement in laminar flows for PCMs and offered a new method for enhancing heat transfer performance by concentrating the PCMs to a layer near the heated wall rather than distributing PCMs homogeneously in the entire channel. Various techniques such as pinched flows [74], inertial lift forces [75], magnetophoresis [76] and acoustophoresis [77] can be used to control the location of the micro/nanoparticles. A two-dimensional model of laminar flow between parallel plates is illustrated in Fig. 8a. At the bottom wall constant heat flux is applied while the top wall is adiabatic. PCMs are confined to a layer of thickness δ near the heated wall. From Fig. 8b, as the PCMs are distributed homogeneously in the channel ($\delta/H=1$), there is a large region with local Nusselt numbers less than that without PCMs ($\delta/H=0$), which degrades the heat transfer. As δ decreases, the deterioration in the local Nusselt number vanishes when $\delta/H \le 0.4$. The average Nusselt number over the entire melting region (Nu_{melt}) has an optimized increase of 30% when $\delta/H=0.3$ compared to that without PCMs, as given in Fig. 8c. Although this investigation is for narrow parallel plate channels, it can provide insights for further heat transfer enhancement of PCMs in micro/minichannels by carefully selecting the PCM particle distribution.

3. Flow boiling enhancement for micro/minichannels

Flow boiling-based cooling, which utilizes the latent heat of vaporization and maintains a relatively uniform temperature distribution, is considered as a promising method to address high heat-flux dissipation needs. Due to the large surface area to volume ratio, flow boiling micro/minichannel heat sinks may allow for very high heat removal rates, giving significant advantages over their macroscale counterparts. Nowadays, thermal design of electronic chips with high heat flux has become one of the major challenges in the next generation electronics. The ongoing miniaturization trend of ICs results in constantly increasing chip-level power densities. The demand for higher computational performance results in microchips with an increasing number of cores and amount of cache memory [78]. Therefore, efficient heat dissipation in three dimensional integrated circuits (ICs) is crucial to support the packing density, reliability and performance scaling of future systems. Two-phase micro-channel cooling is ideally suited for high power electronic chips.

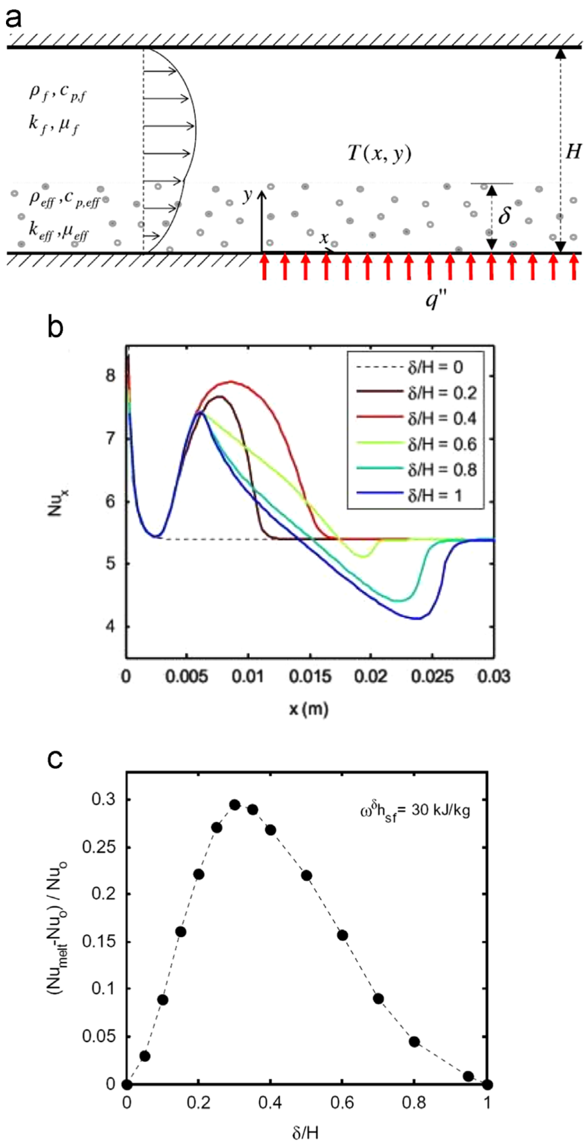

Fig. 8. (a) The schematic model, (b) local Nusselt number and (c) Nusselt number averaged over the melting region (Nu_{melt}) with PCM-particle mass fraction of 20% for δ/H ranging from 0 to 1 compared to the thermally-developed Nusselt number without PCM [57].

Various flow pattern observations are available for micro/minichannels [79–82]. Pressure drop, heat transfer and critical heat flux correlations for flow boiling and condensation for micro/minichannels are proposed in literature [11,83–88]. Most recently, Wu et al. [89] developed relatively simple but accurate flow-pattern based heat transfer predictive correlations for elongated bubbly flow and annular flow in flow boiling micro/minichannels, respectively, based on a collected micro/minichannel database containing 1619 data points for fifteen working fluids. Over 95% of the data points can be predicted by the proposed correlations within a $\pm 50\%$ error band for both elongated bubbly flow and annular flow.

To improve the cooling efficiency of the flow boiling micro/minichannel system, the following four properties are of specific interest: the onset of nucleate boiling (ONB), the heat transfer coefficient (HTC), the pressure drop and the critical heat flux (CHF). CHF represents the operational limit of the heat flux, which occurs at the transition from the nucleate boiling regime or thin film evaporation regime to the film boiling regime—a very poor heat transfer mechanism. Therefore, various heat transfer enhancement techniques are required to (i) decrease the ONB, (ii) increase the HTC, (iii) decrease the pressure drop penalty and (iv) increase the CHF, which will result in higher allowable power densities or, equivalently, more compact cooling systems, with corresponding performance and economic benefits. Bubble dynamics are critical for two-phase flow and heat transfer enhancement in micro/minichannels. Various enhancement techniques are more or less related to bubble dynamics by controlling bubble nucleation, growth, departure diameter, departure frequency and vapor venting to benefit heat transfer enhancement.

3.1. Bubble nucleation

Bubble dynamics is fundamental to the understanding and prediction of boiling heat transfer. Bubble dynamics in microchannels may be quite different from that in conventional channels due to the confined walls. Bubble growth in a microchannel is restrained by the channel wall in the transversal direction and experiences a very large pressure gradient, i.e., shear stress in the streamwise direction [90]. The bubble size is limited by the channel size, in the transversal direction. For bubble departure, buoyancy is negligible while drag should be significant due to the large pressure drop through the channel.

Although the produced bubbles are confined in the micro/minichannels, it is still reasonable to estimate the bubble nucleation by conventional two-phase theories qualitatively, e.g., in Ref. [90]. Hsu [91] obtained a quadratic equation which gives the range of effective nucleating cavities for a given wall superheat as,

$$\{r_{c,\,\min},\ r_{c,\,\max}\} = \frac{\delta_t C_2}{2C_1}\frac{(T_w - T_{sat})}{(T_w - T_{bulk})} \times \left[1 \mp \sqrt{1 - \frac{8C_1 \sigma T_{sat}(T_w - T_{bulk})}{\rho_v h_{lv} \delta_t (T_w - T_{sat})^2}}\right] \tag{2}$$

where $C_1 = 1 + \cos\theta$, $C_2 = \sin\theta$. Assuming a linear temperature profile in the liquid boundary layer, the thermal boundary layer thickness δ_t can be simply expressed as

$$\delta_t = k_l / h_{sp} \tag{3}$$

For thermally fully developed flow, one half of the channel hydraulic diameter can be used as the thermal boundary layer thickness δ_t [90]. Fig. 9a presents the size range of effective cavity sizes for water as a function of wall superheat for different hydraulic diameters and static contact angles. Channels with less hydraulic diameters are relatively more difficult for nucleation. Hydrophobic surfaces with large contact angles seem to have more effective nucleation sizes at lower wall superheats, while hydrophilic surfaces will delay bubble nucleation to higher wall superheats.

Basu et al. [92] found that the active nucleation site density depends only on the static contact angle and wall superheat. The effects of velocity and local liquid subcooling are implicit in the relation between local heat flux and wall superheat and they do not influence the nucleation site density independently. The correlations developed by Basu et al. [92] are given as follows,

$$N_a = 0.34[1 - \cos(\theta)]\Delta T_w^{2.0} \quad \Delta T_{w,\mathrm{ONB}} < \Delta T_w < 15^{\circ} \tag{4}$$

$$N_a = 3.4 \times 10^{-5}[1 - \cos(\theta)]\Delta T_w^{5.3} \quad \Delta T_w \geq 15^{\circ} \tag{5}$$

Fig. 9b illustrates the nucleation site density as a function of wall superheat and static contact angle. Larger nucleation site density will be given for higher contact angles. Therefore, low wetting surfaces with high contact angles tend to have a larger effective cavity size range and a larger nucleation site density. In other words, the wall superheat at ONB for low wetting surfaces

will be lower than that for high wetting surfaces and more bubbles will grow on low wetting surfaces. Low wall superheat at ONB can mitigate flow instability while more bubbles may enhance heat transfer. Therefore, micro/minichannels with low wetting walls may give a higher HTC than that with high wetting walls, especially in the bubbly flow pattern at very low vapor qualities. However, intermittent dryout can appear very easily for low wetting surfaces. The produced bubbles collapse easily and form a vapor layer near the wall and prevent liquid rewetting. The HTC will decrease sharply with a low CHF. On the contrary, high

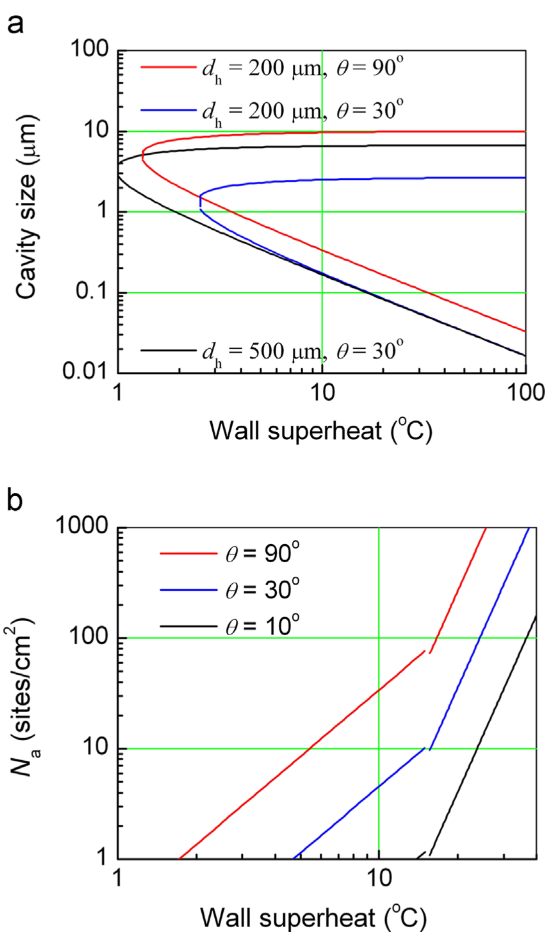

Fig. 9. (a) Effective nucleation cavity size range and (b) nucleation site density versus wall superheat.

wetting surfaces can promote liquid rewetting and therefore delay CHF. Due to the above advantages and disadvantages of low and high wetting surfaces, patterned surfaces with both of them may be very promising in decreasing the wall superheat at ONB, increasing HTC and CHF. For example, hydrophilic/hydrophobic patterned surfaces, e.g., arrangement of hydrophobic spots patterned on a background hydrophilic surface, have been developed to enhance pool boiling HTC and CHF simultaneously [93,94]. For flow boiling in micro/minichannels, performance of such combined patterned walls needs to be investigated in the future.

3.2. Flow boiling enhancement techniques

Several typical flow boiling enhancement techniques in micro/minichannels are given in Table 3. Most of these enhancement concepts for micro/minichannels have been studied over the past five years, such as nanoscale coating and tapered manifold. Generally, area increase, surface characteristics (e.g., porosity, microcavities, wettability), flow instability mitigation by manipulating bubble dynamics, vapor venting and flow uniformity are the main enhancement mechanisms. The four main characteristics, i.e., wall superheat at ONB, pressure drop penalty, HTC and CHF are covered, as listed in Table 3. It is sometimes hard to achieve a definite conclusion for each characteristic of each technique. For example, surface roughness can enhance heat transfer with an accompanied pressure drop penalty. However, asymmetric nano-pillars proposed by Chu et al. [129] can be deposited on the micro/minichannel surfaces to achieve unidirectional bubble movement, and thus may give less pressure drop compared to that of the smooth micro/minichannel at the same working conditions. Thin porous coating will benefit heat transfer while thick coating will present an additional thermal resistance for heat transfer and may deteriorate HTC. Consequently, optimizations should be performed for those enhancement techniques for different applications or working conditions. Other techniques such as interrupted microfins [130], seed bubbles [131], self-sustained high frequency oscillations [132] and jet impingement [133] which are not listed in Table 3, can also enhance flow boiling in micro/minichannels.

Challenges in flow boiling in micro/minichannels are the high pressure drops and flow reversals induced by rapidly expanding confined and elongated bubbles, leading to large pressure fluctuations and flow instabilities and thereby low HTC and CHF [106]. Flow boiling instability in micro/mini-channels is a very notable problem if improperly addressed. Flow boiling instabilities can seriously modify the flow hydrodynamics, introduce transient surface temperature surges, hinder the thermal performance,

Table 3
Enhancement techniques for flow boiling in micro/minichannels.

Typical techniques	Main mechanisms	Main characteristics[a]
Inlet restriction [95–98]	Flow instability mitigation	$\Delta T_{w,ONB}$ –; ΔP ↑; HTC ↑; premature CHF ↑
Reentrant cavity [99–102]	Intensified nucleation activity; stable bubble nucleation process; flow instability mitigation	$\Delta T_{w,ONB}$ ↓; ΔP –; HTC –; premature CHF ↑
Expanding channels [103–107]	Flow instability mitigation	$\Delta T_{w,ONB}$ –; ΔP ↓; HTC ↑; CHF –
Vapor venting [108–110]	Flow instability mitigation; quality reduction; flow pattern change	$\Delta T_{w,ONB}$ –; ΔP ↓; HTC ↑; CHF –
Tapered manifold [111]	Vapor removal from the heater surface and liquid supply to the nucleation sites; flow instability mitigation	$\Delta T_{w,ONB}$ –; ΔP ↓; HTC ↑; CHF ↑
Nanofluid [112–114]	Wettability change due to particle deposition	$\Delta T_{w,ONB}$ ↑; ΔP ↑; HTC ↕; CHF ↑
Surface modification		
Surface roughness [115,116]	Area increase; flow instability mitigation	$\Delta T_{w,ONB}$ –; ΔP ↕; HTC ↑; CHF ↑
microscale coating [117–119]	More microcavities; flow instability mitigation; area increase; wettability change	$\Delta T_{w,ONB}$ ↓; ΔP ↑; HTC ↕; CHF ↑
nanoscale coating [120–128]	Forming possible microcavities; flow instability mitigation; wettability change	$\Delta T_{w,ONB}$ ↕; ΔP ↑; HTC ↕; CHF ↑

[a] ↑ Increase;↓ decrease;↕ either increase or decrease or no change; – negligible effect or unknown.

generate vibrations, compromise structure integrity, and lead to premature initiation of the CHF. Bergles and Kandlikar [134] argued that all of the CHF studies in microchannels were affected by flow oscillation and stated that the large discrepancies between various microscale HTC data sources might be attributed to flow instabilities. Considering flow instabilities properly and developing methods to mitigate them are key issues for HTC and CHF enhancements. Inlet restrictions shown in Fig. 10a are effective for stabilizing boiling, but they impose a severe pressure penalty on the pump [98]. Artificial nucleation sites, such as reentrant cavities (Fig. 10b) or porous coatings, can mitigate flow instability by decreasing the wall superheat at ONB [99]. The maximum pressure that can be sustained inside the vapor bubble is dictated by the saturation pressure corresponding to the heater surface temperature [101]. Therefore, the reversed flow and flow instabilities developed due to high pressure inside the bubble are suppressed when local wall superheat at ONB is reduced by making more available nucleation sites on the heating surface. The cavity sizes of the artificial nucleation sites can be estimated by Eq. (2) and Fig. 9 to achieve low wall superheat at ONB.

Another approach for flow instability suppression is to adopt expanding microchannels (Fig. 10c), which can also reduce associated pumping cost [106]. In expanding microchannels, bubbles tend to move downstream instead of upstream. For an elongated bubble with an upstream radius of curvature c_u and a downstream radius of curvature c_d, the net surface tension force per unit area (τ) due to channel expansion is,

$$\tau = 2\sigma\left(\frac{1}{c_u} - \frac{1}{c_d}\right) \tag{6}$$

As $c_u < c_d$ for expanding microchannels, the sign of τ is positive and the net surface tension will accelerate bubble removal and mitigate flow instability. Fig. 11 displayed the surface temperature variations for both straight microchannels and expanding

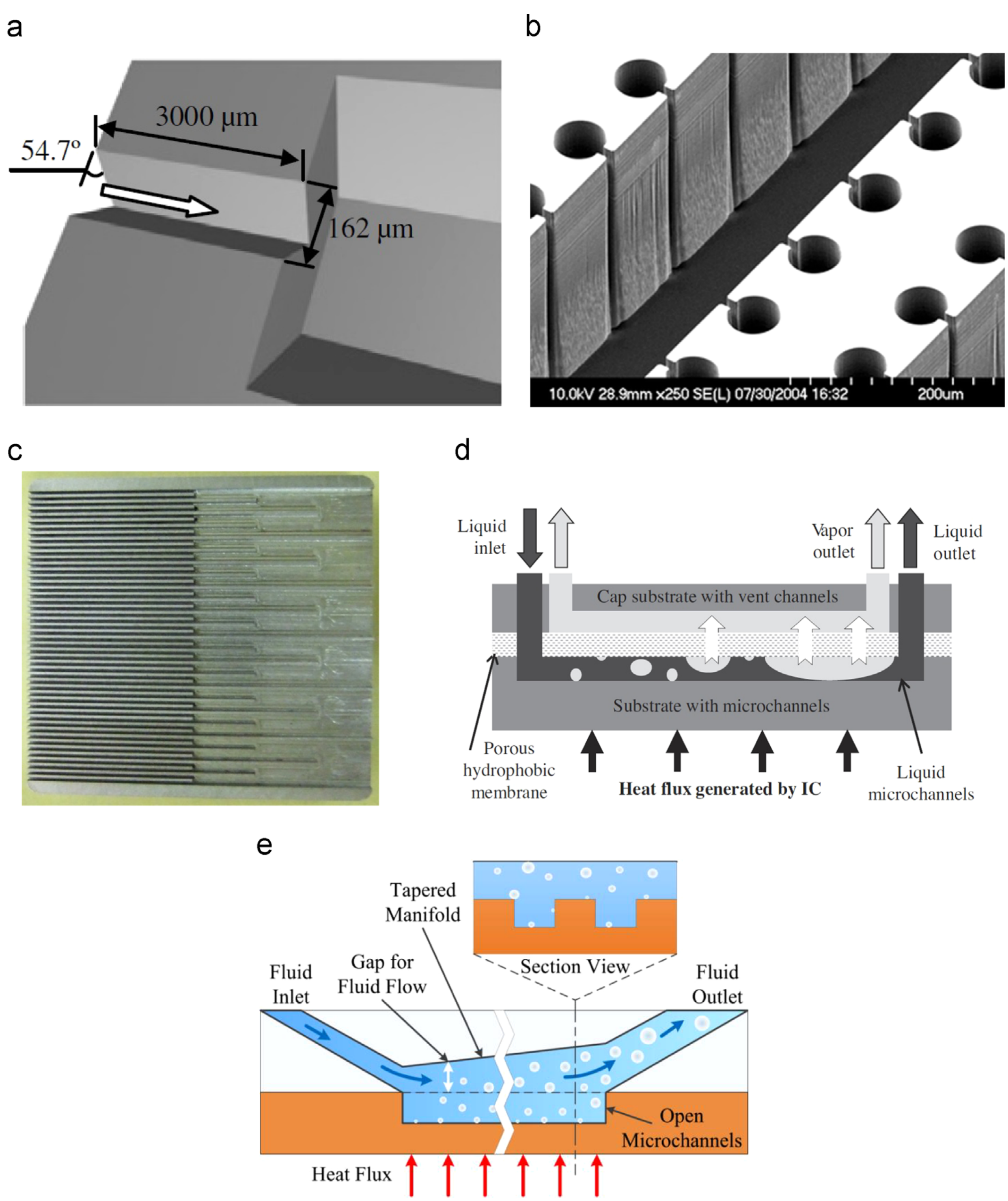

Fig. 10. Schematics of (a) inlet restriction [98], (b) reentrant cavities [99], (c) expanding microchannel array [106], (d) vapor venting by using a porous hydrophobic membrane [108] and (e) open microchannel with tapered manifolds [111].

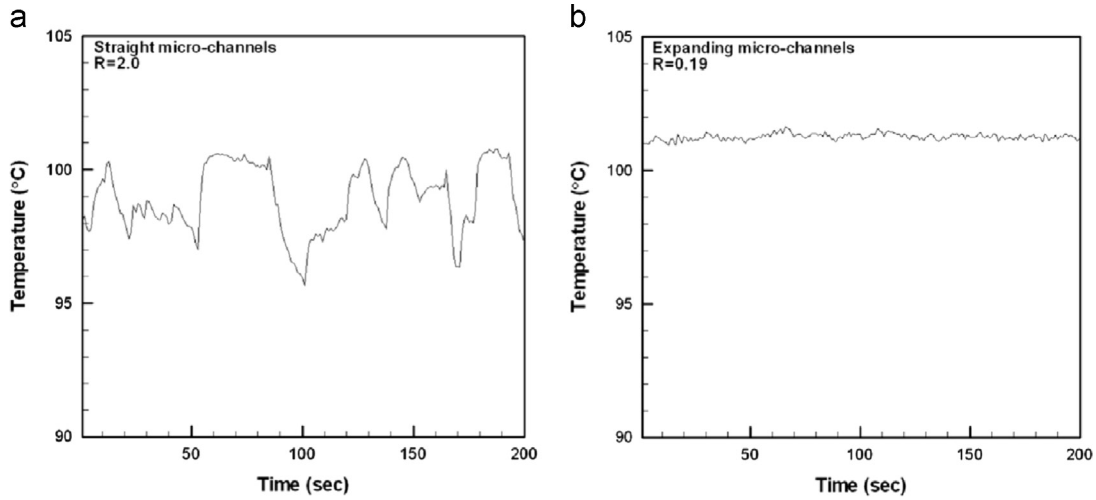

Fig. 11. Surface temperature variations for both straight microchannels and expanding microchannels at the same operating conditions [106].

microchannels at the same operating conditions. In Fig. 11, *R* is the instability parameter established by Lee et al. [106], representing the ratio of the backward force and the forward force acting on the bubble. *R* should be less than unity to achieve a stable flow in evaporating microchannels. As shown in Fig. 11, the surface temperature for expanding microchannels is much more stable than that of straight microchannels. Balasubramanian et al. [107] experimentally investigated flow boiling pressure drop and heat transfer characteristics in straight and expanding microchannels. Fig. 12 indicates that the pressure drop of expanding microchannels is significantly lower and the HTC is higher for expanding microchannels than that for straight microchannels at high heat fluxes. Therefore, the advantage of expanding microchannels is evident.

Another promising method to decrease flow instability, decrease pressure drop and increase HTC is to use vapor venting membranes on the upper sides of the micro/mini-channel array for vapor escape into separate vapor venting channels. This method is designed specifically for water–vapor two-phase system. Yao et al. [135] first proposed to use vapor venting microchannels for microscale fuel cell development. The vapor venting microchannel has a great potential for high heat flux or high vapor generation systems. This is because a microchannel has large surface area but small cross-section so that vapor has no place to go. Venting gas resolves this basic restriction. The porous membrane is the key component for vapor venting and enables the separation and transport of the vapor phase. The basic theories of selecting or fabricating the porous membrane in terms of pore size, porosity, intrinsic permeability, membrane thickness, wettability, and thermo-mechanical stability decide the venting performance. The ideal membrane for use in flow boiling micro/mini-channel heat sinks would: (a) posses a larger leakage pressure than the maximum operating pressure along the membrane and (b) has a low pressure drop across the membrane to improve the vapor removal rate [136]. Fig. 10d gives an example of vapor venting by using a porous hydrophobic membrane. There are three major components: the two-phase microchannels, the vapor venting channels and the porous hydrophobic membrane. The membrane separates the vapor phase from the two-phase mixture and transports it to the venting channel. Fang et al. [109] performed a 3D numerical simulation of the vapor-venting process based on the volume of fluid (VOF) method together with models for interphase mass transfer and capillary force. As shown in Fig. 13a, the pressure drop in vapor venting microchannels initially increases with the heat flux beyond the ONB at a rate similar to

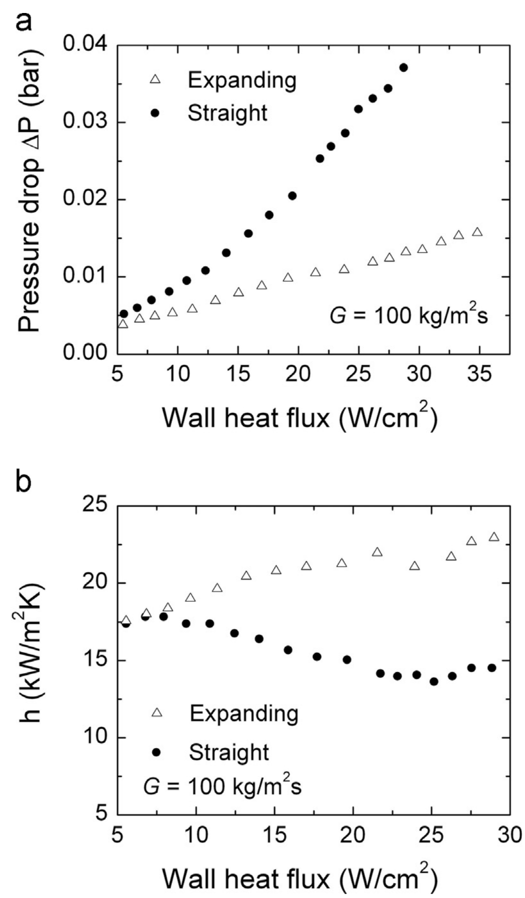

Fig. 12. Performance of (a) pressure drop and (b) HTC versus wall heat flux for expanding and straight microchannels at a mass flux of 100 kg/m^2 s [107].

that of the conventional non-venting microchannels. However, the curve starts to flatten out with a further increase in heat flux beyond around 10 W/cm^2, which clearly shows the advantage of the vapor venting microchannel over conventional microchannel in terms of pressure drop reduction. A 60% reduction in the normalized pressure drop can be seen in Fig. 13a at relatively high heat fluxes. The HTC in the vapor venting microchannels is approximately the same or higher as that in conventional microchannels for a mass flux of 307 kg/m^2 s, as given in Fig. 13b.

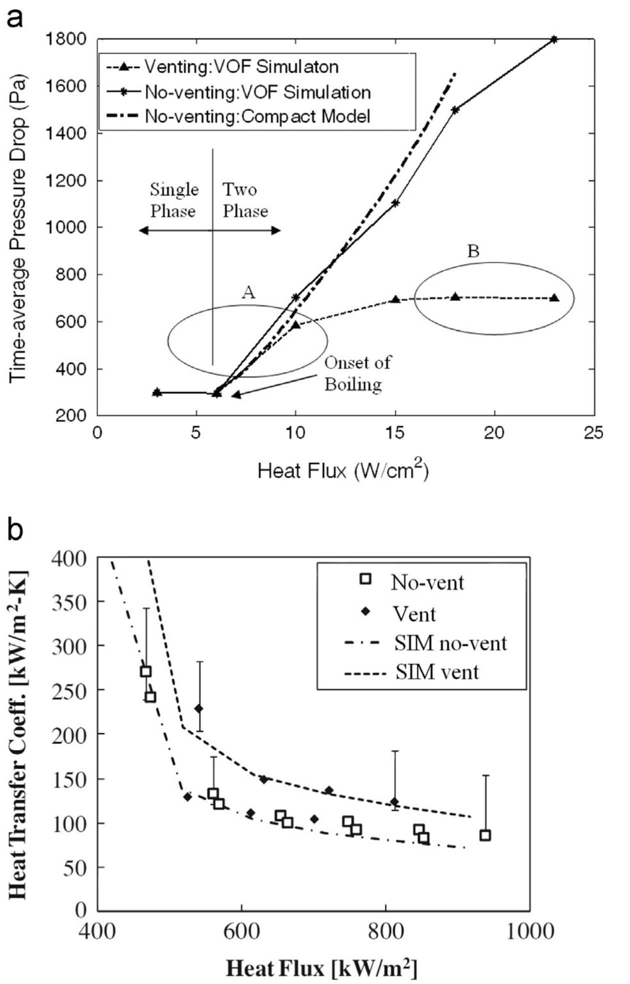

Fig. 13. Performance of (a) pressure drop [109] and (b) HTC [108] versus heat flux for vapor venting and no-venting microchannels.

In addition, up to 4.4 °C reduction in the average substrate temperature was observed for vapor venting microchannels [108]. Optimization of vapor venting microchannel devices should be based on the coupled mechanism between the flow pattern in the microchannel and the flow regime in the vapor venting channel, which needs further study. Very recently, a novel open microchannel design with tapered manifolds (OMM) was developed by Kandlikar et al. [111], as shown in Fig. 10e. Similar to expanding and vapor venting microchannels, OMM has more space for bubbles or vapor and therefore can facilitate bubble removal. The experimental studies confirmed that the OMM design is able to enhance flow boiling HTC and CHF.

3.3. *Micro/nanoscale surface modification*

The effectiveness of the boiling heat transfer phenomenon can be greatly enhanced through careful control of the chemico-physical characteristics of the heat transfer surface [137]. In particular, surface roughness (texture), wettability, porosity and porous structure play crucial roles in determining the size and areal density of vapor bubbles as well as the rate at which bubbles leave the surface. For a given fluid, these surface characteristics determine the HTC that is achievable in a particular device, which in turn affects the efficiency and economics of the device. Most investigations presented in the literature so far, concern engineered surfaces focusing on microscale structures such as micro-roughness, micro-cavities and micro-porosity for heat transfer enhancement, especially for pool boiling [138–140]. Generally, microscale engineered surfaces effectively change all surface parameters including surface roughness, wettability and porosity simultaneously. The mechanisms for how microscale engineered surfaces enhance HTC and CHF are still not well understood. Most recently, pool boiling and flow boiling of nanofluids offered inspiring insights into the engineering of boiling surface by nanoscale features. Nanofluids, engineered colloidal suspensions of nanoparticles in a base fluid, have potential to enhance HTC and CHF. Many studies [141–143] have established that the HTC and CHF enhancement by nanofluids results from a thin porous nanoparticle deposition layer on the heater surface which serves to improve the wettability and capillarity of the boiling surface. The main disadvantages of using nanofluids for boiling heat transfer improvement are the nanoparticle pollution and poor nanoparticle deposition strength [144]. The progress in nanoengineering opens new room for nanoscale surface modification without a large change in the surface topography at microscale.

Investigation on nanoscale engineered surfaces for the boiling phenomenon is at its very early stage. The following points can be gained from the previous research, which will guide the design of micro/nanoscale surface modification for flow boiling heat transfer enhancement in micro/minichannels to achieve high heat-flux dissipation and fine temperature control.

- Micro/nano-engineered surfaces, if properly arranged, can enhance boiling heat transfer performance through modification of the chemico-physical characteristics of the heat transfer surface, e.g., surface roughness, wettability, porosity, porous structure etc.
- Hydrophilic/hydrophobic patterned surfaces, e.g., arrangement of hydrophobic spots patterned on a background hydrophilic surface, may enhance HTC and CHF simultaneously. Hydrophilic surfaces can promote liquid rewetting and therefore delay CHF. Hydrophobic surfaces can decrease the ONB and therefore increase the HTC. Parameter optimization of the patterned surfaces is required for further improvement.
- Porosity can improve the heat transfer performance of a boiling surface, largely due to the interconnection of nucleation sites. However, the thickness of the layer also presents an additional thermal resistance to the heat transfer system.

3.3.1. *Nanoscale coating*

Carbon nanotube (CNT) and nanowire coatings deposited on the microchannel walls are given in Fig. 14. A problem for CNT and nanowire coatings is that the CNTs and nanowires can easily deform and bend upon the wall in response to external forces. Flow velocity can cause appreciable changes to the morphology of the CNT-coated surface. For example, SEM images by Khanikar et al. [121] show that the initially near-vertical CNTs (Fig. 14b) were bent upon the heated surface at high flow velocities to form a repeated 'fish-scale' pattern (Fig. 14c) which results in time-dependent HTC and CHF characteristics. Silicon nanowires given in Fig. 14d were synthesized in situ in parallel silicon microchannel arrays to suppress the flow instability and to augment HTC [124]. Significant enhancement in HTC was demonstrated in Fig. 15 at a higher mass flux of 357 kg/m^2 s due to flow instability mitigation and early ONB. However, at a lower mass flux of 119 kg/m^2 s, the HTC of nanowire coated microchannels is inferior to that of plain-surface microchannels, which can be explained by the evolution of the annular flow pattern at different mass fluxes. More detailed discussions can be found in Li et al. [124]. The nanowire-coated coatings can withstand

high mass fluxes and no visible morphology change can be detected for the nanowire coatings, as can be seen in Fig. 14d (before tests) and Fig. 14e (after tests). The microcavities after the tests remain almost identical to those before the tests. Thus, the nanowire coatings in this work have good structural integrity and can enable long-term flow boiling enhancement.

3.4. Future research needs and the concept of flow-pattern based enhancement

Information on bubble dynamics in micro/minichannels is relatively scarce. Bubble dynamics in micro/minichannels need to be investigated experimentally and numerically to better

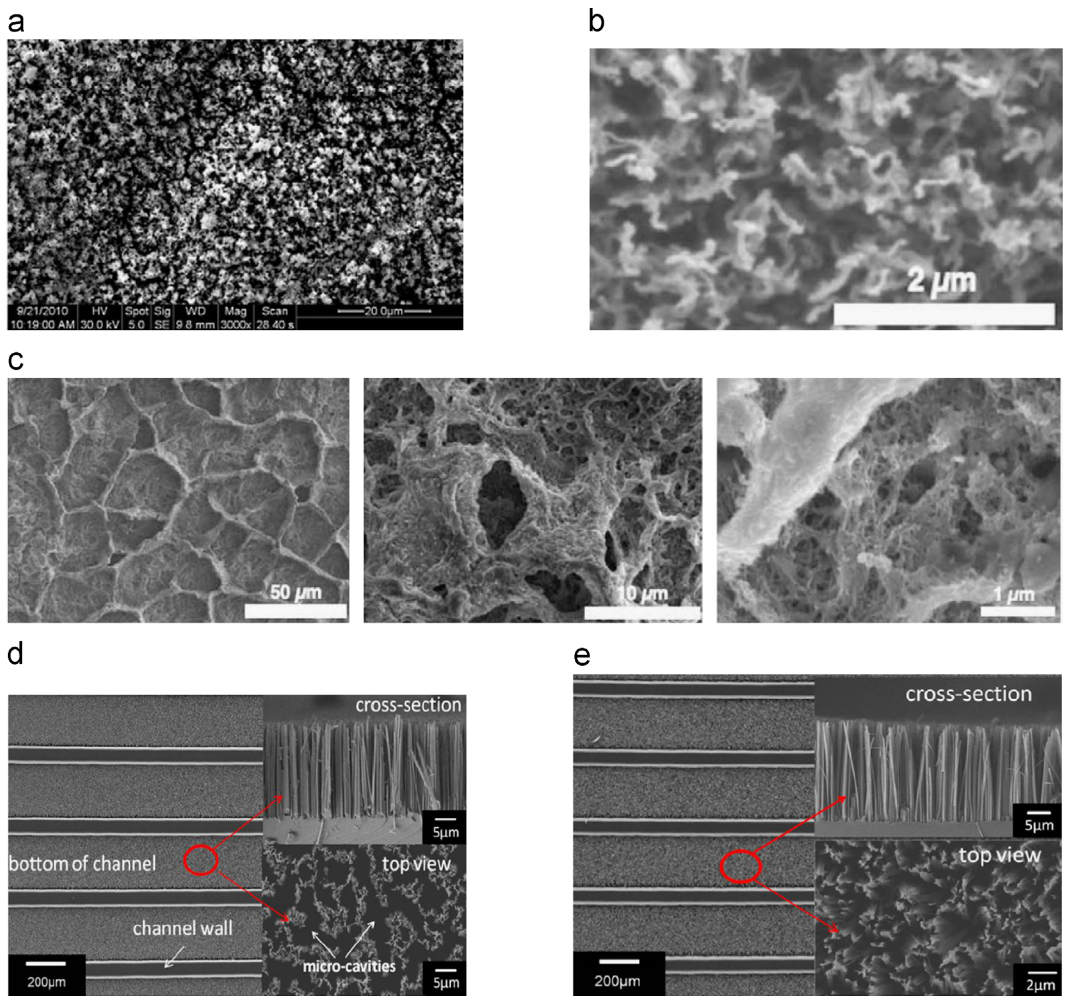

Fig. 14. SEM images of (a) Cu nanowires [120], (b) CNT-coated surface before tests and (c) the same surface after five boiling tests at a mass flux of 368 kg/m^2 s [121], (d) Si nanowire-coated microchannels before flow boiling tests and (e) the same Si coating after flow boiling tests [124].

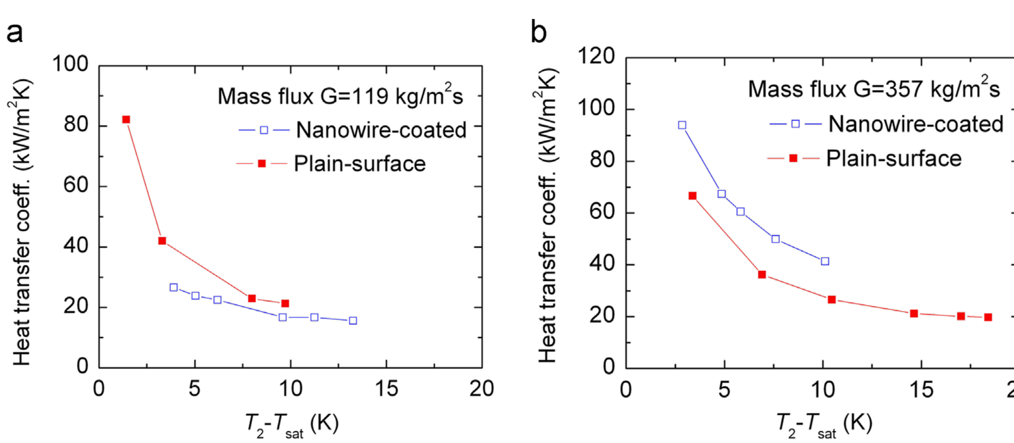

Fig. 15. The local HTC values versus the local wall superheat for the plain-surface microchannels and the nanowire-coated microchannels for mass fluxes of 119 kg/m^2 s and 357 kg/m^2 s [124]. T_2–T_{sat} dictates the local wall superheat.

understand flow boiling behavior in enhanced microchannels (next-generation microchannels) and guide development of new enhancement techniques. Conventional two-phase theories tend to under-predict the wall superheat at ONB. In addition, bubble coalescence, bubble departure and removal are critical for efficient HTC and CHF enhancement. Only with these knowledge, proper arrangement of cavities and their size ranges can be obtained to achieve low flow instability and avoid intermittent dryout during flow boiling.

Most of the flow boiling enhancement techniques have not been proposed for micro/minichannels until very recently. More comprehensive and comparative studies are required. Especially, visualization techniques and local temperature measurements are preferred for flow pattern observation and understanding of enhancement mechanisms. Flow patterns in enhanced micro/minichannels may differ from that in conventional micro/minichannels due to flow instability mitigation and wettability change. As different flow patterns present different heat transfer mechanisms, enhancement techniques should be selected and optimized based on different flow patterns. For example, more effective nucleation sites are beneficial for bubbly flow at very low vapor qualities, while for elongated flow and annular flow, those techniques, e.g., nanostructured roughness and thin porous coatings which can promote thin film evaporation and liquid rewetting, should be adopted to prevent intermittent dryout and therefore enhance HTC and CHF accordingly. Moreover, these enhancement techniques should be optimized based on the different flow patterns. Many flow boiling processes cover large vapor quality changes and different flow patterns, so composite enhancement techniques can be applied for different sections corresponding to different flow patterns along the whole microchannel length.

Combined enhancement techniques may be very promising to develop super-cooling microchannel heat sinks. For example, structural roughness elements can be deposited on the microchannel walls together with the nanowire or CNT coatings. Then the nanowires or CNTs will bend around the roughness elements instead of upon the microchannel walls. The HTC and CHF will probably be enhanced due to the large amount of microcavities formed by the nanowires or CNTs between roughness elements and the superior thermal conductivity of the nanowires or CNTs. The nanowires and CNTs will conduct heat from the bulk liquid to the wall and therefore increase the heat transfer rate. Another example is to use structural roughness elements or even hierarchical roughness to enhance heat transfer. The roughness profiles will affect the wettability. Superhydrophobic or superhydrophilic surfaces can be obtained by hierarchical roughness. Therefore, wettability modification or control is necessary for hierarchical roughness surfaces to prevent dryout or flooding. As flow boiling enhancement in micro/minichannels is very new, neither of the above combined techniques has been studied in the literature. Development of new enhancement techniques can be based on and inspired by progresses in nanotechnology and bioengineering.

Optimal choice over the heat transfer enhancement techniques is of great concern. However, in the choice of a suitable technique for a specific heat transfer equipment or process, both economic and physical factors are important. This requires knowledge of cost effectiveness but this is outside the scope of this work.

4. Concluding remarks

A state-of-the-art overview of the most recent enhancement techniques for further single-phase flow and flow boiling enhancement in micro/minichannels has been presented and relevant future perspectives have been provided in this paper. A special emphasis was on those enhancement techniques with high heat transfer enhancement and relatively low pressure drop penalty. The main concluding remarks for further enhancement of single-phase flow and flow boiling in micro/minichannels are stated as follows.

4.1. *For single-phase flow in micro/minichannels*

- Similar to single-phase enhancement in conventional channels, the main enhancement mechanisms for single-phase flow in micro/minichannels are boundary layer interruption and thinning, repeated developing flow, secondary flow and fluid mixing, heat-transfer area increase, jet impingement, enhanced thermal conductivity and heat capacity, etc.
- Typical single-phase enhancement techniques are listed in Table 2. Enhancement techniques, e.g., dimples/oval dimples, wavy/curved microchannels, supercavitating flow, oblique fins and streamlined fins are promising in enhancing heat transfer with low pressure drop penalty.
- Single-phase HTC can be intensified effectively by replacing the traditional continuous channel walls with offset strip microfins. Fin shapes, fin dimensions and fin array parameters (fin pitch and fin attacking angle) should be optimized together to achieve a trade-off between HTC and pressure loss. Staggered S-shape fins and streamlined fins such as airfoil-shaped fins can achieve the same heat transfer performance as continuous flow passages with a much lower pressure loss.
- The enhanced thermal conductivity of nanofluids is beneficial for HTC, while the associated viscosity increase of nanofluids entails higher pressure loss. No agreement on anomalous heat transfer enhancement of nanofluids has been achieved up to now. Possible effects such as Brownian motion, thermophoresis and diffusiophoresis on heat transfer enhancement in microchannels need to be investigated in the future. Nanofluid stability may still be a problem for long-term use.
- Micro/nano-encapsulated particles of phase change materials (PCM) can be adopted as additives in liquids for heat transfer enhancement due to their high energy storage density and small temperature variation.
- Combined enhancement techniques may be very promising for single-phase heat transfer enhancement and require further research. For example, using PCM slurries and streamlined microfins can achieve a high HTC, a relatively low pressure loss and an almost even surface temperature distribution.

4.2. *For further flow boiling enhancement in micro/minichannels*

- Heat transfer enhancement techniques for flow boiling are required to (i) decrease the wall superheat at ONB, (ii) increase the HTC, (iii) decrease the pressure drop penalty and (iv) increase the CHF, which will result in higher allowable power densities or equivalently, more compact cooling systems, with corresponding performance and economic benefits.
- Bubble dynamics are closely related to flow pattern, pressure drop and heat transfer. Effects of wettability on bubble nucleation are given in Section 3.1 in detail. Low wetting surfaces tend to have a larger effective nucleation cavity size range and a higher nucleation site density, therefore a lower wall superheat at ONB and a higher HTC. However, intermittent dryout can appear easily for low wetting surfaces, even at relatively low vapor qualities. As high wetting surfaces have high CHF, patterned surfaces with proper arrangement of both low and high wetting surfaces may have low wall superheat at ONB and provide appreciable HTC and CHF improvement.
- Flow boiling instabilities in micro/minichannels can seriously modify the flow hydrodynamics, introduce transient surface temperature surges, compromise structure integrity, decrease

HTC and lead to premature CHF. Therefore, flow boiling instability should be properly addressed and reduced to obtain high HTC and CHF values. The flow instability can be mitigated by many techniques including inlet restrictions, artificial nucleation sites such as reentrant cavities or porous coatings, expanding microchannels, vapor venting and OMM design etc.

- Micro/nanoscale coatings or micro/nano-engineered surfaces, can enhance flow boiling HTC and CHF with negligible or low pressure loss through modification of the chemico-physical characteristics of the heat transfer surfaces, e.g., surface roughness, wettability, porosity, porous structure etc.
- Under flow and fierce boiling conditions, the morphology of nanoscale coatings, such as carbon nanotube coatings may change, resulting in time-dependent HTC and CHF performance. Aging performance of the coatings requires further investigation.
- The concept of flow-pattern based heat transfer enhancement for flow boiling has been proposed in this study. Because different flow patterns present different heat transfer mechanisms, enhancement techniques should be selected and optimized based on different flow patterns.
- Combined enhancement techniques may be very promising for development of super-cooling microchannel heat sinks, as detailed in Section 3.4. More experimental investigations should be conducted first to have an overview in this area.
- Enhanced surfaces and porous structures can attract surface contaminant easily due to intermolecular forces, accelerating fouling of the heating surface. Tackling fouling problems has gone hand-in-hand with improved heat transfer. The fouling characteristics of flow boiling in micro/minichannels with enhanced structures have not been covered in the scientific literature yet. Knowledge about fouling characteristics in enhanced micro/minichannels needs to be obtained in the future.

Heat transfer enhancement is an evergreen and important topic of huge relevance in existing, future and renewable energy systems as well as for energy conservation and environment protection. Heat transfer enhancement techniques for micro/mini-channels are very new and far from mature, and call for continued research and development.

Acknowledgements

Financial support from the Swedish National Research Council (project number: 621-2013-3897) and the Swedish Energy Agency is gratefully acknowledged. The authors want to thank Prof. S.C. Yao at Carnegie Mellon and Prof. W. Li at Zhejiang University for valuable discussions.

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